## Citation

## As Published
http://dx.doi.org/10.1109/ITHERM.2010.5501375

## Publisher
Institute of Electrical and Electronics Engineers (IEEE)

## Version
Final published version

## Citable link
http://hdl.handle.net/1721.1/72958

## Terms of Use
Article is made available in accordance with the publisher’s policy and may be subject to US copyright law. Please refer to the publisher’s site for terms of use.
DESIGN AND ANALYSIS OF HIGH-PERFORMANCE AIR-COOLED HEAT EXCHANGER WITH AN INTEGRATED CAPILLARY-PUMPED LOOP HEAT PIPE

Matthew McCarthy¹, Teresa Peters¹, Jon Allison¹, Alonso Espinosa¹, David Jenicek², Arthur Kariya¹, Catherine Koveal¹, John G. Brisson¹, Jeffrey H. Lang², and Evelyn N. Wang¹

¹Department of Mechanical Engineering, ²Department of Electrical Engineering and Computer Science
Massachusetts Institute of Technology
77 Massachusetts Avenue
Cambridge, MA, USA, 02139
Phone: (617) 253-6401
Email: mattmcc@mit.edu

ABSTRACT

We report the design and analysis of a high-power air-cooled heat exchanger capable of dissipating over 1000 W with 33 W of input electrical power and an overall thermal resistance of less than 0.05 K/W. The novelty of the design combines the blower and heat sink into an integrated compact unit (4” x 4” x 4”) to maximize the heat transfer area and reduce the required airflow rates and power. The device consists of multiple impeller blades interdigitated with parallel-plate condensers of a capillary-pumped loop heat pipe. The impellers are supported on a common shaft and powered with a low-profile permanent magnet synchronous motor, while a single flat-plate evaporator is connected to the heat load.

KEY WORDS: heat exchanger, air cooling, capillary-pumped loop heat pipe, permanent-magnet motor, wick

NOMENCLATURE

A          winding cross sectional area, m²
B          magnetic flux density, T
d          motor thickness, m
g          grav. acc., m/s²
gap, m
I          current, A
k          number pole pairs
L          length, m
m          number windings
N          number active phases
P          pressure, Pa
p          power, W
R          radius, m
t          thickness, m
T          temperature, K
w          width, m
z          distance, m

Greek symbols

γ          surface tension, N/m
ρ          mass density, kg/m³
ρₑ         electrical resistivity, Ohm/m
θ          contact angle, radians
τ          torque, Nm
ω          rotational speed, rad/s

INTRODUCTION

Thermal management is a critical bottleneck for high-power systems, such as phased-array radar and microwave and digital electronics, where performance and reliability are dictated by the ability to dissipate heat efficiently. Fluidic-based cooling solutions have commonly been incorporated using traditional large-scale air-cooled fin-fan arrays and pumped liquid-based cooling [1-3], but have suffered from system inefficiencies and limited performance characteristics.

The current work focuses on developing a novel high-power cooling unit comprised of a complex capillary-pumped loop heat pipe with an integrated blower. The pump-heat sink (Figure 1) is 4” x 4” x 4” and consumes less than 33 W of electrical power while dissipating over 1000 W of heat with a thermal resistance of less than 0.05 °C/W. The design consists of a series of rotating blades used to force air between parallel condenser plates (~2.5 mm thick) of the capillary-pumped loop heat pipe with a single evaporator layer at the bottom.

Fig. 1: Schematic of the 4” x 4” x 4” integrated pump-heat sink device.
The integrated design utilizes a stack of rotating blades interdigitated between 18 thermal stator plates, each of which is a condenser chamber of a capillary-pumped loop heat pipe. A single evaporator layer is located at the bottom of the unit, which is in contact with the heat source, and connected to the thermal stator plates via vertical pipes. A low-profile radial-flux permanent magnet motor is mounted on top and drives the rotors on a single shaft running through the condenser plates. Air enters the top through an axial intake and is drawn radially outward between the condenser plates. The successful operation of the heat pipe relies on efficient wicking of the water through the complex stacked geometry. The heat pipe is an important component of the system needed to create a near isothermal heat sink and to minimize the overall thermal resistance. Figure 2 shows a schematic of the cross-section of the device and Figure 3 shows a schematic detailing the condenser and evaporator wicking structures. The working fluid (water) evaporates in the primary evaporator wick; the vapor travels through vertical pipes to the condenser layers where it is convectively cooled to the liquid phase and then wicked back into the evaporator. Distinct wick sizes are necessary to create high driving pressures in the evaporator and low permeabilities in the fluid transport sections.

**INTEGRATED BLOWER DESIGN**

While the capillary-pumped loop heat pipe creates a near isothermal sink for heat removal, the radial outflow blower convectively cools the stack of condenser layers. The interdigitated nature of the impeller design creates high heat transfer rates through two mechanisms. First, the developing flow results in high heat transfer coefficients directly at the heat transfer surface. Secondly, the rotating blades shear off the developing boundary layer on the condenser surfaces as they pass by. Both of these mechanisms are derived from the direct integration of the fan and fin components in this design. Typical air-cooled heat exchanger systems have separate blowers ducted to the heat sink. Accordingly, this integration is a key component of the proposed design.

Figure 4 shows first order results for system thermal resistance and air flow rate as well as the motor torque and power necessary to drive the impeller blade stack. These results are obtained from iteratively solving closed form analytical models of convective heat transfer coefficient of developing flows between parallel plates [4] and the Euler turbine equation. This simplified modeling approach assumes an isothermal surface temperature of 75 °C and no flow deviations from a general impeller blade design with a swept angle of 45°. While it does not capture the complexities of the turbomachinery flow field, this model has been used as a means of establishing a baseline design, which will be refined through the use of computation fluid dynamics (CFD). This preliminary analysis results in total of 18 interdigitated impeller blades with flow paths (condenser to condenser spacing) and blade thicknesses on the order of 2mm.

Using this preliminary design, a three-dimensional rotating-frame CFD model was developed in Ansys Fluent. Figure 5 shows computational domain and representative results of...
pathlines and surface heat flux contours. Using CFD the preliminary design space was investigated through characterization of various design aspects including the number of blades, the ratio of flow field gap to blade thickness, blade angle, and eye radius. For a given geometry, the CFD model was solved by increasing the rotational speed until the required mechanical power reaches 27 W. The mechanical power is computed as the product of the angular speed and the total viscous torque acting on the blades. Based on the design goals, there is 33 W of power available to drive the system and the choice of 27 W is a conservative value with

![Image](image1.png)

Fig. 5 Ansys Fluent (a) computational domain and representative results for (2) pathlines and (b) surface heat flux contours

![Image](image2.png)

Fig. 6 CFD results of thermal power, angular velocity, air flow rate, and viscous torque as a function of (a) blade angle, (b) gap ratio, (c) number of blades, and (d) eye radius.
Table 1. Blower impeller design specifications

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat Dissipated</td>
<td>1250 W</td>
</tr>
<tr>
<td>Mechanical Power</td>
<td>27 W</td>
</tr>
<tr>
<td>Stator Surface T</td>
<td>75 °C</td>
</tr>
<tr>
<td>Rotor Speed</td>
<td>4975 rpm</td>
</tr>
<tr>
<td>Tip Speed</td>
<td>26 m/s</td>
</tr>
<tr>
<td>Eye Radius</td>
<td>20 mm</td>
</tr>
<tr>
<td>Number of Layers</td>
<td>15</td>
</tr>
<tr>
<td>Number of Blades</td>
<td>5</td>
</tr>
<tr>
<td>Blade Sweep at Tip</td>
<td>45°</td>
</tr>
<tr>
<td>Flow Channel Thickness</td>
<td>2.5 mm</td>
</tr>
<tr>
<td>Rotor Thickness</td>
<td>1.5 mm</td>
</tr>
<tr>
<td>Gap Thickness</td>
<td>500 μm</td>
</tr>
<tr>
<td>Stator Thickness</td>
<td>2.5 mm</td>
</tr>
</tbody>
</table>

up to 6 W available for motor and bearings losses. The thermal power (heat removal), viscous torque, air mass flow rate, and angular velocity were calculated for each geometry. Figure 6 shows how these values vary with several geometric parameters. The blade sweep, eye radius, and blade number have a small effect on the thermal power removed for the range of geometries considered. Accordingly, the final blower design was selected based on this analysis as well as manufacturing constraints; the number of layers was decreased to fifteen, as compared to the preliminary closed form model, with a flow channel thickness of 2.5 mm and gap thickness of 0.5 mm. Accordingly, this numerical simulation and optimization procedure resulted in a design that not only meets the heat removal (>1000 W) and mechanical power (<30 W) requirements but also reduces manufacturing complexities. Table 1 shows the final impeller blade design based on these CFD results.

**LOOP HEAT PIPE ANALYSIS**

Loop heat pipes utilize the evaporation and condensation of fluid to transfer heat. Capillary forces develop in porous wicks and drive flow between an evaporator and, traditionally, a single condenser. Figure 7 shows a schematic of the current loop heat pipe design with multiple flat plate condensers stacked in parallel with a single evaporator. The condenser sections each contain wicking structures to completely separate the liquid and vapor phases. This is necessary to be insensitive to orientation in a gravitational field and to keep higher condenser layers from flooding the lower layers. The analysis of loop heat pipes has been well documented with design guidelines for the basic operating principles clearly defined [5]. This work focuses on the design requirements arising from the use of parallel condensers with wick structures in a stacked geometry. Specifically, the need to lower the liquid-side temperature and the creation of a two-phase compensation chamber insensitive to orientation is addressed.

While the wick in Figure 7 is drawn as a plug in the condenser flow path, the actual condenser geometry will have porous wick covering the top and bottom surfaces of the parallel plate.

Fig. 7 Schematic of loop heat pipe operation showing evaporator wick, parallel condenser sections with wicks, liquid compensation chamber, and liquid and vapor lines.

Fig. 8 Pressure-temperature diagram of the capillary-pumped loop heat pipe operated upright and inverted under gravity forces, where the blue curve represents the saturation line.
vapor space. This ensures that as vapor condenses, it is immediately wicked into the liquid space and condensation will occur uniformly over the upper and lower surfaces. As liquid evaporates at the surface of the evaporator wick, a meniscus forms in the porous material generating a net pressure rise that balances the total pressure drop through the system. Similarly, another meniscus forms as liquid condenses in the condenser plates and wicks in the porous surface. For stable robust operation insensitive to gravitational effects and orientation, it is desirable to have receded wicks in the condenser were the meniscus curves inward toward the wick. To achieve this, the pressure on the liquid side of the device has to be lower than the vapor side.

Figures 7 and 8 illustrate the steady state operating cycle of the loop heat pipe oriented upright or inverted with respect to gravity. Liquid evaporates across the evaporator wick surface from state 8 to state 1 with the appropriate pressure rise to drive the loop. The maximum capillary pressure rise is

\[ \Delta P_{8-1}^{\text{Max}} = 2\gamma \cos \theta / R_E \]  

(1)

where \( \gamma \) is the surface tension of water, \( \theta \) is the contact angle between the water and the wick, and \( R_E \) is the pore radius of the wick. The vapor travels through the heated evaporator section to state 2 where the temperature and pressure drops to a superheated state. The vapor travels through the vapor lines and into each parallel condenser section to state 3. The prime notation in Figure 8 denotes the condenser closest to the evaporator and the un-prime notation is for the condenser farthest from the evaporator. For sufficiently large pipe diameters (>0.25" inner diameter) the viscous losses in the vapor lines are negligible and can be ignored, therefore 3 and 3' are at the same point. The pressure drops across the receded condenser meniscus as the vapor condenses from state 3 to 4. The liquid sub-cools and loses pressure as it flows through the porous wick to state 5 and continues to cool through the liquid passages of the condenser to state 6. For the upright orientation, the liquid pressure increase due to gravitational head as it flows downward to the compensation chamber in state 7. For the inverted case, the liquid pressure will drop as the liquid flows up to state 7. Viscous losses in the open liquid sections will be negligible due to the low volume flow rates but the temperature will increase as the liquid flows to the compensation chamber. Liquid is then wicked into the evaporator wick and drops pressure and increases temperature as it flows to state 8. From there the liquid evaporates across the evaporator meniscus and the cycle begins again.

State 7 is the two-phase compensation chamber containing primarily liquid but with a small vapor bubble, achieved by under-filling the device with water. The condenser and evaporator wicks will suppress the formation of bubbles within them and because the compensation chamber is the hottest location in the open liquid side, the vapor will preferentially form there. Because the compensation chamber is two-phase, the pressure will be set by the saturation pressure and therefore temperature. Accordingly, by controlling the heat leak through the evaporator wick, the compensation chamber temperature can be reduced leading to a lowered liquid side pressure.

By having wicking structures and a stable meniscus in the condenser sections the vapor phase and liquid phase are completely separated. This allows the liquid side pressure to vary without affecting the vapor side. Figure 8 shows the effects of orientation on the liquid side pressure, where the states 4-6 (and 4'-6') are shifted depending on the position of the compensation chamber relative to the condensers. In both cases the liquid side compensation chamber pressure (state 7) is pinned by the saturation temperature of the vapor bubble and the vapor pressure in the condensers (state 3) is pinned by the evaporator pressure. The condenser meniscuses adjust as the distribution of gravitational head in the liquid side is varied.

For successful operation of this device, the condenser meniscuses must curve inward for stable separation of the two phases eliminating flooding and/or dry-out. This is due to limitations in the manufacturability of defect free surfaces as well as the stability of advancing meniscuses under dynamic thermal and mechanical loading. For the condenser meniscuses to recede the vapor pressure at state 3 must exceed the liquid pressure at state 4 (or state 4').

\[ P_3 > P_4 \]

(2)

The vapor pressure in the condenser is given by

\[ P_3 = P_1 - \Delta P_{1-3}^{\text{vis}} \]

(3)

where \( P_1 \) is the evaporator pressure and \( \Delta P_{1-3}^{\text{vis}} \) is the viscous pressure drop between the evaporator and condenser and has a positive value. The liquid pressure at the condenser wick surface is

\[ P_4 = P_7 - \rho g z + \Delta P_{4-5}^{\text{vis}} \]

(4)

where \( \rho \) is the water density, \( g \) is the acceleration of gravity, \( z \) is the distance above the evaporator for a specific condenser, and \( \Delta P_{4-5}^{\text{vis}} \) is the viscous pressure drop through the condenser wick given by Darcy’s law [5] and has a positive value. Combining equations 2-4 and rearranging gives

\[ P_1 - P_7 > \Delta P_{4-5}^{\text{vis}} - \Delta P_{1-3}^{\text{vis}} - \rho g z \]

(5)

The pressure difference across the evaporator wick can be given by

\[ P_1 - P_7 = (dP/dT)(T_1 - T_7) \]

(6)

where \( dP/dT \) is the slope of the saturation curve at the operating temperature. Combining equations 5 and 6 yields the thermal criteria for the compensation chamber
where \( z \) can vary from negative to positive values of up to 4 inches based on the desired size of the device. This inequality must be met to keep the liquid side of the device at a low enough pressure so that the condenser meniscuses are always receded, ensuring stable operation. Additionally, the condenser wick must be able to provide the sufficient capillary pressure to maintain the pressure differences across it. The maximum pressure that can be supported across the condenser wick is

\[
\Delta P_{\text{Max}}^{\text{3-4}} = 2\gamma \cos \theta / R_C
\]

(8)

where \( R_C \) is the pore radius of the condenser wick. Combining equations 3, 4, 6, and 8 yields the criteria for supporting the condenser meniscus

\[
2\gamma \cos \theta / R_C > (dP/dT)(T_1 - T_7) + \rho g z - (\Delta P_{\text{vis}}^{\text{3-5}} + \Delta P_{\text{vis}}^{\text{4-5}})
\]

(9)

Similarly by relating equations 1 and 6, the evaporator wick must meet the criteria

\[
2\gamma \cos \theta / R_E > (dP/dT)(T_1 - T_7) + \Delta P_{\text{vis}}^{\text{7-8}}
\]

(10)

where \( \Delta P_{\text{vis}}^{\text{7-8}} \) is the viscous pressure drop through the evaporator wick given by Darcy’s law [5]. Equations 7, 9 and 10 outline the thermal and fluidic criteria need to maintain stable meniscuses and device operation of the proposed design. The criteria relates the properties of the wicking structures to the achievable temperature difference across the evaporator, the viscous losses in the liquid and vapor lines as well as the effects of random orientation in a gravitational field.

While continued work focuses on implementing this modeling approach for the design and manufacturing of the parallel plate loop heat pipe, the preliminary design of the evaporator and condenser sections can be seen in Figures 9 and 10. The evaporator consists of a multi-layered wicking structure fabricated in a stainless steel frame and sandwiched between two copper plates. The thermal criteria required for stable operation (Eq. (7)) is achieved by using a low-thermal-conductivity stainless steel primary wick. By incorporating an insulating wick material, the two-phase compensation chamber temperature will be lower than the vapor side of the evaporator. This temperature difference will pin the liquid side pressure (defined by the saturation pressure of the compensation chamber vapor bubble) to a value below that of the vapor side of the device. A secondary coarse copper wick is used to provide adequate fluid distribution across the evaporator area. The assembly is brazed together using a silver/copper brazing alloy.

Figure 10 shows a schematic representation of the condenser geometry and assembly procedure. Chemically-etched stainless steel plates are filled with a large-pore sintered-metal wick and brazed together in a stacked plate geometry. By incorporating wick over the entirety of the vapor chamber uniform condensation (and therefore uniform temperature and heat flux) can be achieved. The vapor and liquid regions are separated using an intermediate spacer to define an in-plane sub-cooling section. The multi-layered stack will be assembled by brazing vertical fluid tubes through the various plate geometries.

Fig. 9 Exploded and cross-sectional views of the proposed evaporator design showing the planar multi-layered wick for thermal isolation of the integrated two-phase compensation chamber.

Fig. 10 Exploded and cross-sectional views of the proposed condenser design showing the sub-cooling section and planar wicking geometry.
LOW-PROFILE MOTOR

The integrated air-cooled heat exchanger shown in Figure 1 is driven using a low-profile permanent-magnet synchronous motor mounted onto the top of the device. The motor is designed to deliver 30 W of mechanical power at the design speed of 5000 rpm (Table 1) with a total electromagnetic loss of less than 3 W. Figure 11 shows a schematic of the motor detailing the relevant components. The motor must fit into the 4” x 4” square footprint of the device and 10 cm has been allocated for its thickness. The motor design and optimization process is as follows. First, the magnetic flux ($B$) of a given geometry is determined by solving Laplace’s equation for magnetic potential across the radial gap between the copper windings and the permanent magnets (Figure 12). From there the total current ($I$) is calculated from

$$\tau = \frac{p}{\omega} = I \cdot B \cdot d \cdot R_m$$  \hspace{1cm} (11)

where $\tau$ is the torque, $p$ is the mechanical power, $\omega$ is the angular speed, $d$ is the depth of the motor, and $R_m$ is the radius of the rotor-stator gap. The current flowing through a single winding is given by

$$I_w = \frac{I}{2Nmk}$$  \hspace{1cm} (12)

where $N$ is the number of active phases, $m$ the number of windings per pole per phase, and $k$ the number of pole pairs. The Ohmic loss for a single winding is given by

$$p_{\text{loss}} = I_w^2 \rho_e L / A$$  \hspace{1cm} (13)

where $\rho_e$ is the electrical resistivity, $L$ is the length, and $A$ is the cross sectional area of an individual winding. From this analysis, the total Ohmic loss of the motor can be calculated based on the number of windings. The motor optimization has been carried out by repeatedly solving these equations using an algorithm implemented in MATLAB. The geometric parameters have been systematically varied with the magnetic flux solved and the losses calculated for each permutation.

---

Fig. 11 Schematic of the synchronous motor, showing the rotor core, stator core, copper windings, and magnets.

Fig. 12 Magnetic flux field for a given geometry.

Fig. 13 (a) Optimization results for Ohmic losses as a function of number of pole pairs and motor thickness. (b-c) Geometric variables and the optimized design values.
The results of this optimization procedure was filtered based on the manufacturability of the design and the minimum losses encountered. Figure 13 shows representative results of the analysis showing the effects of the number of pole pairs and motor thickness on losses as well as the various geometric variables and the resulting values optimized through the described method. With this design, less than 1 W of Ohmic losses and 0.5 W of electromagnetic losses have been predicted for mechanical power output of up to 30 W. Along with an assumed mechanical loss of ~ 1W from the bearings, this design will meet the criteria for 90% efficiency.

CONCLUSION

The design and analysis of an air-cooled heat exchanger comprised of an interdigitated array of blower impeller blades and thermal stators has been presented. A radial outflow impeller design has been optimized through computational fluid dynamics for enhanced air-cooling in a parallel-plate geometry. The modeling of two-phase loop heat pipes with flat plate condenser geometries has been presented with design criteria for stable operation with condenser wicks. A low-profile permanent-magnet synchronous motor has been numerically optimized for efficient operation at the design speed of the integrated device. These results demonstrate the viability of the proposed integrated air-cooled heat exchanger and will act as the basis for the design and fabrication of such devices.

ACKNOWLEDGEMENTS

This work is supported by the Defense Advanced Research Projects Agency (DARPA) Microsystems Technology Office (MTO) Microtechnologies for Air-Cooled Exchangers (MACE) program, Grant Number W31P4Q-09-1-0007, with Dr. Tom Kenny as the program manager.

REFERENCES